Performance of REBAM® during ball bearing failures

Bently Nevada Corporation has, for the last decade, offered a monitoring system for machines with rolling element bearings called REBAM®, an acronym for Rolling Element Bearing Activity Monitor. This system uses a high-gain proximity probe to observe small deflections in the bearing outer ring as elements under load pass over the probe location. Bearing failure testing, using REBAM®, was originally performed using bearings with artificial flaws (grooves machined into the raceway), but no testing had been performed with natural fatigue spalling. Recently, a research facility and test rig were developed to gain additional knowledge about the performance of REBAM® during the bearing failure process (fatigue spalling) and the effects of bearing loading on REBAM® signals. This required designing a testing apparatus which would allow sufficient loading of a bearing to facilitate natural fatigue spalling in a relatively short time, while preventing the test bearing from overheating.

Figure 1 shows the conceptual design of the test rig. Loading of the test bearing is achieved hydraulically with independent radial and axial loading cylinders. Radial loading is achieved with a hydraulic ram bolted to the side of the casing which directly loads a cylindrical roller bearing mounted in a floating housing, in turn causing loading of the test bearing. By mechanical advantage, the test bearing sees approximately twice the amount of radial load as that input by the radial cylinder. Axial loading is achieved by an axial loading piston mounted in an annular cylinder which is machined into the inboard endplate. This load is transferred to the shaft via a cylindrical roller thrust bearing and taken up by the test bearing. Loading can be controlled between zero and 5000 lbs (22.2 kN) radially and zero and 6000 lbs (26.7 kN) axially.

The test bearing was instrumented (Figure 2) with four REBAM® probes mounted 90 degrees apart around the circumference of the outer ring to allow simultaneous data acquisition from multiple probe orientations. The probes were identified as Channels 1 through 4, with Channel 1 being nearest (11 degrees clockwise from) the center of the radial load zone and the other three numbered sequentially in the direction

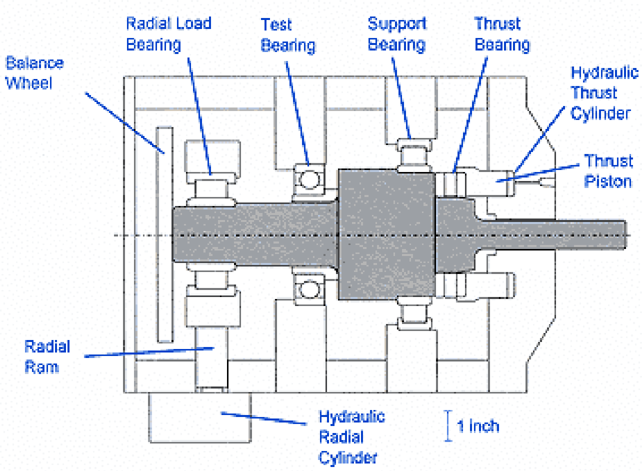

Figure 1
REBAM® test rig.

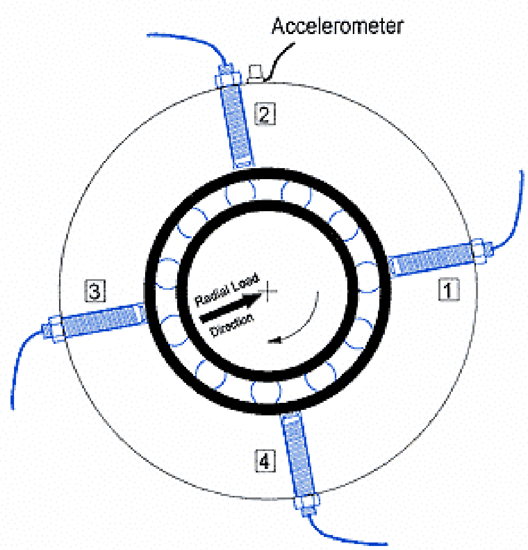

Figure 2
Test bearing transducer layout.

opposite shaft rotation (shaft rotation was clockwise). In addition to the REBAM® transducers, a high-frequency accelerometer (4.5 - 14000 Hz ±3 dB) was mounted on the external surface of the test bearing housing. The accelerometer was mounted near true vertical orientation. From a vibration transmissibility standpoint, the accelerometer was optimally mounted. Actual machinery usually has material gaps, webbing, and longer distances between the source of vibration (bearing) and the transducer that the vibration must traverse. This causes attenuation of the vibration signal. The data from the accelerometer was therefore taken in a "best case scenario."

For the initial bearing failure testing, a 214 Conrad-type deep groove ball bearing was used for the test bearing. The operating speed was set at 3550 rpm. For the test bearing, at this speed, the bearing characteristic frequencies were calculated as:

Outer Race Element Pass Component = 270 Hz
Inner Race Element Pass Component = 381 Hz
Cage Rotational Component = 24 Hz
2X Element Spin Component = 336 Hz

There are two frequency bands which have been the traditional regions of interest with the REBAM® system: The Rotor Frequency Region, which is defined as ¼X to 3X shaft speed, has been used to monitor vibration components originating from rotor-related sources (unbalance, misalignment, instability, etc.); and the Prime Spike Region, defined as the frequency band between the Outer Race Element Pass frequency and seven times this value (denoted 1Ep$_x$ - 7Ep$_x$), has traditionally been used to detect bearing flaws. We will see, however, with natural fatigue spalling we can actually detect significant action in the rotor region when inner race and/or element spalling is present.

Bearing failure under radial load

REBAM® testing included running a bearing to failure under radial loading and again under axial loading. This article will discuss the radial loading data; the next issue of *Orbit* will discuss the axial loading data.

The first objective in testing was to collect some baseline data on a new bearing for later comparison with signals from a damaged bearing. Various combinations of radial and axial loads were placed on the test bearing, and the corresponding signals were recorded on tape. Under pure radial loading, the baseline signal amplitude from the Channel 1 REBAM® probe increased linearly with applied load. Channels 2 and 4 also increased relatively linearly, but with a lower slope value. The major component of these signals consisted of the outer race element pass component (270 Hz). The Channel 3 signal decreased with applied load. This side of the bearing was being unloaded as radial load was increased.

Following the baseline data testing, a fatigue spalling failure was achieved under radial loading. There was some difficulty in keeping the test bearing temperatures below an acceptable limit (trip level was set at 215°F, 102°C), so the amount of loading on the test bearing was varied somewhat during testing. The radial loading during this test averaged about 3000 lbs (13.3 kN). A very small, half-moon shaped pit 0.040 inch long and 0.020 inch wide was placed in the outer raceway near the location of the Channel 1 probe by electrostatic discharge to provide a location of stress concentration for fatigue spalling to initiate. This initial hole was small enough so that no evidence of it was seen on the instrumentation (REBAM® or seismic).

After approximately 20 hours of continuous operation, evidence of spalling could be seen on the Channel 1 REBAM signal (see Figure 3). This was seen as a small negative spike in the Timebase waveform which occurred once every element pass cycle. In the frequency domain, this manifested itself as an increase in the amplitudes of the harmonics of the Outer Race Element Pass frequency (2EPx, 3EPx, 4EPx, etc.). The 1EP$_x$ component was probably also increased, but under this excessive loading it was already very high, making the change less noticeable. After 29 hours of operation, the bearing was inspected for damage progress. The damage was limited to the spalling in the outer ring (Figure 4). This spalling occurred in the outer race very near the Channel 1 probe (initiated by the small hole). There was no visible spalling on either the elements or the inner race at this point.

Testing was then resumed under 3000 lbs (13.3 kN) of radial load. After about 35 hours (total run time), the Channel 2►

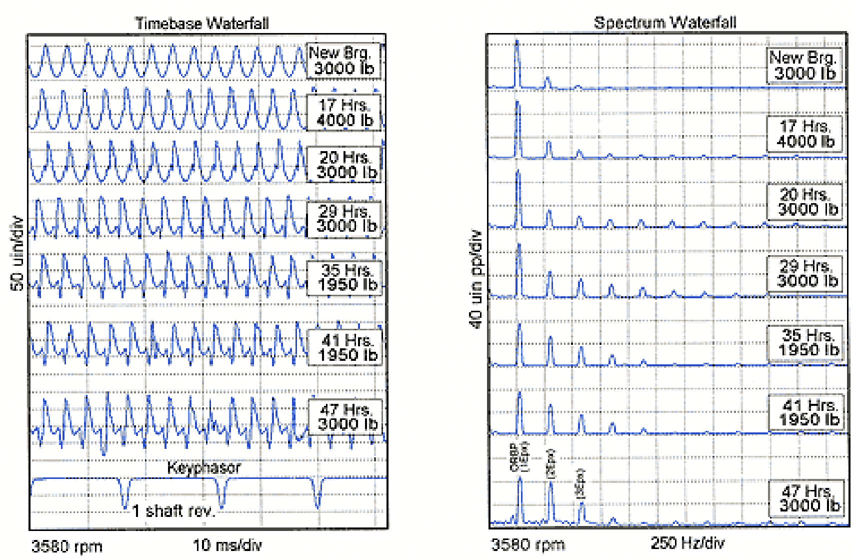

Figure 3
Failure progression under radial loading as seen from Channel 1 REBAM® Probe (11 degrees from center of load zone).

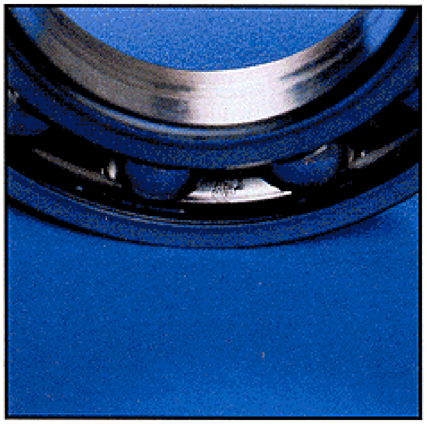

Figure 4
Evidence of spalling in the outer ring.

and Channel 4 REBAM® signals also started showing evidence of spalling. It did not show up in the same fashion on Channels 2 and 4 as it did on Channel 1. Instead of a once-per-element-pass negative spike, Channels 2 and 4 started showing erratic behavior with negative spiking which occurred once-per-shaft-revolution. This 1X spiking was more easily viewed in the live oscilloscope data than the data shown in Figure 5. In the frequency domain, this showed as increased amplitudes in the Prime Spike region *and Rotor region*. Channel 3 was in a severely unloaded location with this extreme radial load and therefore did not show much amplitude before or after spalling. This does not happen under realistic loading, as we shall see. The failure progression as seen on the Channel 2 REBAM® probe is shown in the Timebase and Spectrum Waterfall plots of Figure 5.

To understand why the deflection characteristics for Channels 2 and 4 are different from Channel 1 we need to understand the forces involved. The amount of loading on a particular element dictates how much the outer ring will deflect underneath it. A spalled area is a location where material has flaked away due to fatigue. When an element is in a position where spalling exists at its inner race or outer race contact points, the element is temporarily unloaded since greater clearance exists at this instant. The load given up by this element is taken up by the surrounding elements. At the instant this element is unloaded, the ring is allowed to "spring back" to its unloaded condition. If a probe is near this location, it will show this as a negative spike in its signal. Channel 1 was seeing this every time an element passed its location since there was spalling in the outer race near its location. Channels 2 and 4 only saw this when an inner race and/or element spall coincided with their locations. For inner race spalling, this was once-per-shaft-revolution. Element spalling would show up unpredictably since the element is free to twist, causing the flaw to avoid contact.

Figure 6 shows the failure progression as seen from the accelerometer. After 17 hours under 3000 lbs (13.3 kN) radial load (when the Channel 1 REBAM® signal started to show damage), the accelerometer signal was already significantly higher in amplitude than the baseline level. The first three samples shown were taken with a bandwidth of 2500 Hz and the remaining samples at 5000 Hz. As bearing damage progressed, the impulse excitation caused by elements impacting

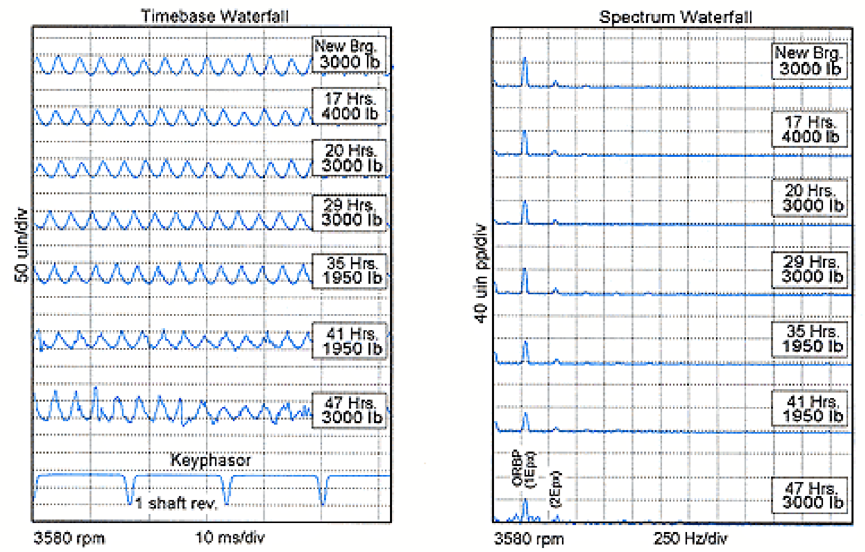

Figure 5
Failure progression under radial loading as seen from Channel 2 REBAM® Probe (79 degrees from center of load zone).

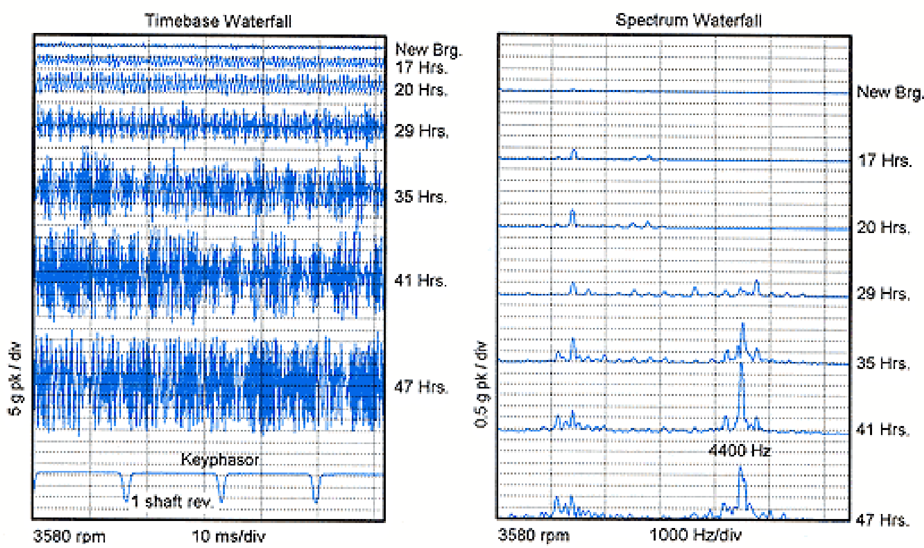

Figure 6
Failure progression under radial loading as seen from accelerometer.

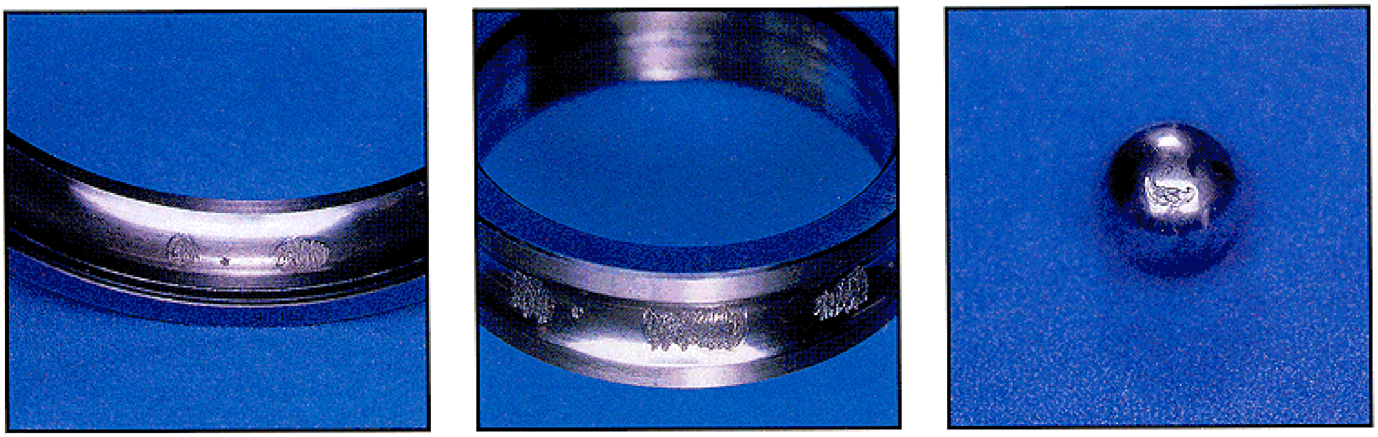

Figure 7
Final spalled condition of the bearing (from left to right: Outer, Inner, Element).

spalled areas excited structural resonances in the system. The first resonance was at about 1400 Hz, and the second (which could be seen in the Spectrum after the recording bandwidth was changed to 5000 Hz) is seen at about 4400 Hz. Transducer resonance was in excess of 50 kHz, and was therefore not a factor. Very little was seen in the accelerometer signal at any of the bearing characteristic frequencies. These components were likely below the noise level of the signal.

After failure was achieved on the test bearing (when the rig could no longer run without the monitoring system tripping it off-line), the bearing was inspected again. Significant spalling was seen on the inner race, and one of the elements had a small, half-moon shaped pit. Figure 7 shows the final spalled condition.

It is apparent, after reviewing the data, that the accelerometer provided an earlier signal level amplitude increase than the REBAM® signals. If this were a field installation, however, a neighboring bearing or machine could be causing the vibration, making it difficult to determine which bearing (or even which machine) the vibration was coming from. Since the REBAM® system measures ring deflections *relative to the housing*, it is not sensitive to structural vibrations and, therefore, provides a much clearer picture of the selected bearing's condition.

Failure detection under realistic radial loading

It also should be kept in mind that very high loads were used to fail this bearing. As such, the baseline REBAM signal levels were much higher than normal. This is due to the much greater loading on each element deflecting the ring over the probe to a greater degree than they would under normal field loading. More advanced spalling conditions are then necessary to show significant changes in the REBAM® signals above the baseline levels. We would therefore expect a greater percentage increase in signal level under field-level loading conditions for a given amount of spalling.

With this in mind, the next step was to use a bearing with extensive spalling and place it under low radial load (300 lbs, 1.33 kN). This was the bearing that was run to failure in the **axial** load failure test (the data from this failure progression will be discussed in the next issue of *Orbit* in part II of this article). It contained spalling on the inner raceway and several of the elements. The signal levels were then compared with the baseline levels at the same load. The speed was again 3550 rpm. Figure 8 shows the signal from the Channel 1 REBAM® probe for the baseline and failed bearings. The signal from the Channel 1 probe showed a 130 percent increase in overall signal level, Chan-►

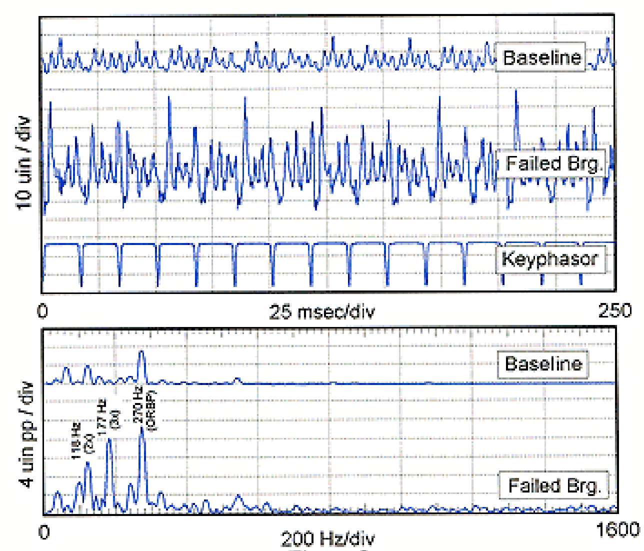

Figure 8
Baseline and failed bearing data under 300 lbs. (1.33 kN) radial load as seen from Channel 1 REBAM® Probe.

nel 2 showed a 160 percent increase, Channel 3 showed a remarkable 860 percent increase, and Channel 4 increased by 214 percent. The large increase seen on Channel 3 is due to its small baseline amplitude. This shows one very important point: If a REBAM probe is mounted directly opposite the center of the radial load zone in a bearing under pure radial loading, you may not see a large signal when the bearing is healthy or new, but when the bearing fails, it will likely produce enough deflection in the outer ring to produce a large change in signal amplitude. Much of the change occurred at the outer race element pass frequency, but we also see a large change in the components in the rotor frequency region, particularly the 2X and 3X components.

Conclusion

This testing under radial loading has allowed us to draw some important conclusions about the operation and performance of the REBAM® system:

1. REBAM® probes which are mounted away from an outer race spall will likely show evidence of bearing damage when the spalling progresses to the inner race or elements.
2. The REBAM® system is best utilized as an indicator of when a bearing needs to be replaced. A seismic transducer (casing mounted accelerometer) may give earlier warning of bearing wear but may also cause the user to replace the bearing before its economic life is over.
3. The REBAM® system gives a much clearer picture of what is occurring in a particular bearing than a seismic transducer.
4. A REBAM® probe which is mounted in the least optimum angular position (180 degrees from the radial load zone center) is still likely to give very good evidence of spalling in the bearing.

In the next issue of *Orbit* we will present the data and conclusions from the axial loading failure testing and low axial load testing. ■

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Synopsis Of Saves

The following actual incidents briefly illustrate some of the benefits customers have received using Bently Nevada products in a variety of applications:

An ethylene plant had a problem with the third stage compressor on one of their units. Due to the process, the third stage runs hot and causes a carbon buildup on the turbine blades. Using Dynamic Data Manager®, rotating specialists and operations personnel can monitor amplitude and phase of the third stage bearings on the unit. It allows them to efficiently schedule a compressor wash to remove the carbon buildup.

Through an on-line look at trends from their System 64, a utility customer detected a thrust wear problem on a boiler feed pump. Accelerated wear was only evident after viewing a one-year trend. The discovery allowed investigation of a potential problem that, otherwise, would have been overlooked.

Trendmaster® 2000 allowed a utility company to identify two cooling tower fan gearboxes in need of rework prior to returning them to service. The customer believes this finding saved more than enough to cover the entire cost of the system.

Our Machinery Diagnostic Services group worked with a refining customer to solve structural/rotor resonance problems on a fan. 1X (synchronous) vibration plots from ADRE®3 were used to identify the problem. The client was then able to rectify the situation.

Another utility company effectively used their ADRE® 3 System to successfully diagnose a turbine problem. As the turbine approached half running speed, vibration levels dramatically increased. After analysing the data, it was decided the best alternative was to shut the machine down for inspection. The problem was believed to be a bow in the shaft.

When four turbine bearings had high temperatures, Dynamic Data Manager® DC Gap trend data confirmed the prognosis of wiped bearings. It also indicated that not as many bearings were wiped as originally thought. The utility company saved disassembly time and effort.

Following a motor coupling failure on a compressor machine train, a refining customer was able to determine the cause using a TorXimitor®. It was found that a torsional resonance had been excited, producing high enough torque to induce fatigue failure in the coupling. The increase in torque only occurred within a narrow speed range. The anti-surge control is being redesigned so the unit does not operate within this range.

A 3300 System installed in December 1991 on a scrubber fan provided a high frequency blade pass alarm when the process fouled the fan. The customer experienced a similar problem earlier in the year on the same machine which caused the machine to come apart.

A recently-installed Transient Data Manager™ System at a utility has already proven its value. Following a forced trip, the customer was able to do a balance-correction in one day on a 625 MW unit. This saved at least two days of outage (and associated lost revenue). ■